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No. 824.

A PROBLEM IN CONTINUOUS ROPE DRIVING.

BY SPENCER MILLER, M. Am. Soc. C. E.

PRESENTED DECEMBER 15TH, 1897.

WITH DISCUSSION.

Much valuable literature has already been given to the engineering world on the subject of rope driving, and it is not the author's intention to review the subject generally, but merely to call attention to the perplexing problems which occur in driving with ropes, where the driving and driven sheaves are of unequal diameters, and especially where the driving wheel is the larger; as, for example, in the dynamo drive. It is acknowledged that in such drives, as usually installed, where the ropes are applied as one continuous rope with tension carriage, they will not pull equally, and, as a result, a few of them do the major part of the work, and consequently it is now recognized that a factor of safety of from 30 to 40 is necessarily employed to cover the inequality of tension, imperfections in mechanism, weakness due to grease, splice, etc.

The author is satisfied that while, by the usual method of figuring, a factor of safety of 30 apparently is employed, he may be able to prove that frequently the actual factor of safety is not more than 15.

In considering the problem of rope driving, it is well to bear in mind the following undisputed facts :

First.—For efficiency in rope driving, it is absolutely necessary to increase the driving power of the rope by making angular grooves in the pulleys, the common practice being to make the angle 45 degrees.

Second.—The driving power of the rope increases rapidly as the arc of contact is increased ; for example, the relative power of an arc of 120° is only 75% of that represented by 180 degrees.

Third.—The greater the angle of incline of the grooves, the less will be the wear of the ropes ; the power wasted due to the pulling out of the rope from the groove will also be reduced. This waste, however, is not as great as it is usually considered.

Hence, in designing a rope drive of the character under consideration, the first thing to determine is the sharpest angle of groove permissible for efficiency and wear of rope ; assume, then, that 45° is to be adopted as the minimum angle of groove, especially as this accords with the judgment of a majority of engineers.

Given, therefore, the smaller pulley with its smaller arc of contact grooved to 45°, the larger pulley may with its greater arc of contact have a greater angle of groove, without losing driving effect, and therein is a saving, for with the wider angle of groove, the rope wears less, and there is less power wasted due to the pulling of the rope out of the groove.

In a transmission of this character, if the arc of contact of the small pulley is 150°, and of the large pulley 210°, and assuming a coefficient of friction of .12, the angle of the small pulley being 45°, then the proper angle for the large pulley would be about *65° for equal adhesion on both wheels. Such transmissions are of very common occurrence.

Assuming, therefore, that the ropes are uniform throughout, it is found that :

By widening the groove of the larger pulley, the waste of power may be minimized, the life of the rope prolonged, and at no additional expense, either for sheaves or loss of driving effect.

The author will now attempt to prove that using the wider angle in the larger pulley with the continuous rope plan of driving, and with

* See Table No. 5, page 176.

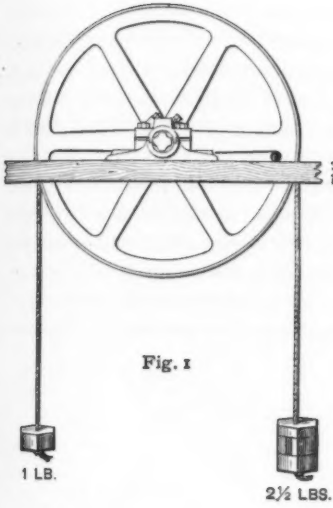

Fig. 1

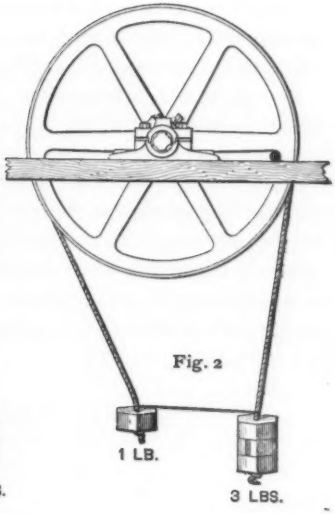

Fig. 2

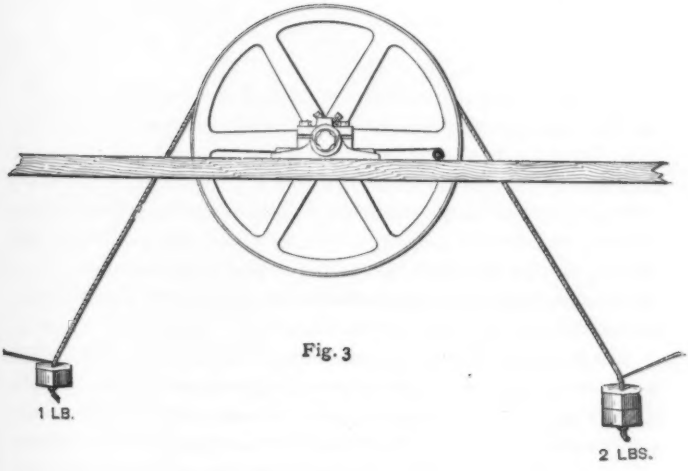

Fig. 3

the same strain on the driving rope, there is a large increase of power-transmitting capacity.

Before proceeding, it is well to review the elements, or the elementary part of rope driving, so as to clearly fix in one's mind exactly what takes place in each of the ropes in driving.

For instance, in imagination, try a few simple experiments: Mount a sheave wheel suitable for rope driving, grooved to 45°, so that it cannot revolve; place in the groove a short piece of rope having on one end, say, a 10-lb. weight; then put on the other end weights until the rope is just ready to slip (see Fig. 1). It may be that the 10-lb. weight will support 25 or 30 lbs., or even 50 lbs., in proportion to the coefficient of friction. The arc of contact in this case is 180°, or one-half the circumference. Assume—for it answers all the purposes of this discussion—that 10 lbs. support 25 lbs.; in other words, the ratio is as 10 : 25, or as 1 : 2½.

Fig. 4.

A series of experiments made by the author several years ago gave for new rope and new sheaves (iron) as high a ratio as 1:7, while with old rope and polished sheaves as low as 1:2½; all wheels grooved to 45 degrees.

If now (by the use of a cord) the ropes are drawn together, so as to embrace, say, 220° of contact, it will be found that the 10 lbs. will support 30 lbs. (see Fig. 2). In other words, the ratio is as 1 : 3. Then try the third experiment, that of diminishing the arc of contact, say, to 140°, and the ratio becomes 1 to 2 (see Fig. 3).

In a rope drive, the part of the rope representing that supporting the 10 lbs. is the slack strand, while that supporting the greater weight is the driving side, or the tight strand of the rope. The tendency of the heavy weight is to revolve the wheel in its direction, and the measure of its effective pull is found by subtracting the pull on the tight side from the pull on the slack side; in other words, as in the first experi-

ment, where the arc of contact is 180°, the tendency of the 10 lbs. is to pull in one direction, while that of the 25 lbs. is to pull in the other direction, and the effective pull is $25 - 10 = 15$ lbs.

The fourth experiment will be to place side by side two sheaves of different diameters, having the same form of groove. Assume that one is 72 ins., and the other 36 ins. in diameter.

If, now, the first experiment is tried with each of these, it will be noticed that 10 lbs. supports 25 lbs., whether the ropes be applied to the 72-in. or the 36-in. sheave. These experiments show that the arc of contact is an important factor in designing a rope drive, and is independent of the diameter of the pulley.

The next matter of importance is to determine what takes place when motion is given to a rope drive.

Fig. 4 shows the simplest kind of a transmission, that of two

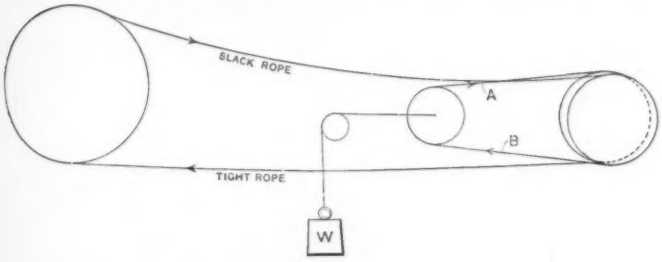

FIG. 5.

grooved wheels and a single rope wrapped about them. When motion takes place and a load is given to the driven wheel, the lower strand straightens up and becomes taut, and the upper strand takes the slack from the driver in the direction in which the rope travels.

If the slack strand is sufficiently heavy, and is drawn up sufficiently taut, it will do the required amount of work. If it is desired to increase the work done by this rope, its tension must be increased. Suppose, therefore, that a loose sheave is added to the side of the driven wheel, and that a traveling idler, held back by a constant weight, as shown in Fig. 5, is put in, then it will be seen that the rope A is the slack rope, governing the tight rope (the lower strand), and the amount of tension permissible on the tight rope is determined by the tension on the rope A.

If the weight on the tension carriage is 150 lbs., then that on the rope A is 75 lbs. (neglecting friction), and with the ratio of slack to tight of 1 to 2, the pull on the tight rope will be 150 lbs. The tight rope will, therefore, throw its slack around the large pulley to the upper strand, which in turn will be at once taken out by the rope B, whose tension is constantly 75 lbs. This tension in the slack strand is more than sufficient to sustain the tight rope, because, on account of the large arc of contact in the driver, the ratio is, say, 1 to 3. Hence, the rope will not transmit more power than the small pulley is capable of taking, and the slack is taken out by the tension carriage in the direction in which the rope travels.

It is worthy of notice that the slack strand governs only the amount which may be pulled by the large wheel, while the tension

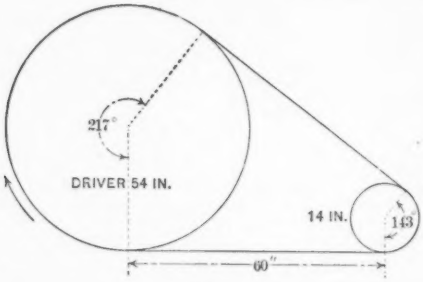

FIG. 6.

rope A governs what the tight rope may pull on the circumference of the small or driven wheel.

It is to be supposed that the reader is fully aware of all the points which have been reviewed, and they are only given here to direct his mind to further thought in this direction. An example in actual practice will be used to assist in further explanation.

A rope drive (Fig. 6) installed a few years ago had a driver which was 54 ins. diameter, and made 320 revolutions a minute; the driven was 14 ins. diameter, and made about 1 200 revolutions a minute. There were five strands of $\frac{3}{4}$-in. rope. Both wheels were grooved to 45°, and the distance between centers was 60 ins. By laying this out, it is found that the arc of contact on the large wheel is 217°, while on the smaller one it is 143° (see Fig. 6). This plant, after running a year,

proved to be a failure, because, as the engineer said, "the rope would break frequently." He said, as he looked at the drive, he could see that "one or two ropes were doing the whole work."

Fig. 7 shows the way in which the ropes were led. Assume a tension weight of 300 lbs. Taking a coefficient of friction of .12, the coefficient of resistance to slipping in the 45° angle is .31.* Taking into account the arc of contact, the ratio of the slack to the tight side is therefore 1 to 2.12 on the small wheel, and 1 to 3,12 on the large wheel. For the sake of easy calculation omit the fraction and consider a traction ratio of 1 to 3, and 1 to 2, respectively, in the large and small pulleys, and omit the weight of rope. Now imagine the engine start-

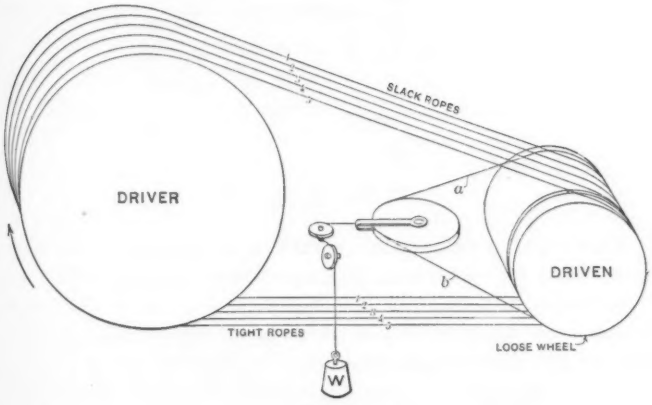

FIG. 7.

ing, a full load of 50 H.-P. demanded on the part of the driven wheel, and trace what takes place. The rope *a* is held under a constant tension of 150 lbs. (half tension weight). This will sustain a pull of 300 lbs. on tight rope *1*, and the first effect of its taking such a tension is to elongate and throw its slack up around large pulley to slack rope *1*. Meantime tight rope *2* has been doing its best, and it will pull out the slack thrown into slack rope *1* to the extent of its ability, viz., in the ratio of 2 to 1. As soon as tight rope *1* is satisfied, as it is with a

* (Coef. of resistance to slipping = Cosec $\frac{\text{angle}}{2}$ × coef. of friction.) See Table No. 6, p. 177.

slack tension of 100 lbs., then slipping ceases in groove *1* of the large pulley, and this 100 lbs. allows tight rope *2* to pull only 200 lbs.

Following the same line of reasoning, the results in the following table are obtained (the slack passing out in the direction of the travel of the rope until the tension carriage takes it up).

TABLE No. 1.—Approximate Strains in the Various Ropes While Transmitting 53 + H.-P., Neglecting Friction Losses, Weight of Rope, and Centrifugal Force. Both Wheels Grooved to 45° Angle:

Tight rope *1*—300 lbs.		Slack rope *1*—100 lbs.	
" *2*—200 "		" *2*— 66 "	
" *3*—133 "		" *3*— 44 "	
" *4*— 88 "		" *4*— 30 "	
" *5*— 60 "		" *5*—150 "	
	781 lbs.		390 lbs.
	390 "		
	391 lbs. effective pull.		

Tight rope *5*, given as 60 lbs., may be raised somewhat by the influence of slack rope *5*, given as 150 lbs., as will be explained further on.

By taking the sum of all these tight ropes, a total of 781 lbs. is obtained, and the sum of all the slack ropes is 390 lbs., making a difference of 391 lbs. of useful work.

$$\frac{391 \text{ lbs.} \times 4\,518 \text{ ft. per min.}}{33\,000} = 53 + \text{H.-P.}$$

The author makes no claim to exactness in these figures; they are only used as a means of demonstrating that inequality of tensions does exist, but do not show the exact amount of such inequality.

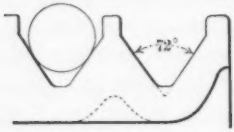

Fig. 8.

Assume the ultimate strength of $\frac{3}{4}$-in. rope to be 4 200 lbs., and in this transmission there is a maximum strain of 300 lbs., which gives an actual factor of safety of only 14.

If now the large wheel (Fig. 8) be grooved to 72°, so as to make the ratio 1 to 2 in each wheel, and the tension weight be reduced to 150

lbs., the result will be perfectly plain by again tracing the strains in the rope.

If a tension weight of 150 lbs. on the tension carriage be assumed, then there would be 75 lbs. tension on the rope *a*, which would give 150 lbs. tension to tight rope *1*, which cannot be sustained until there is 75 lbs. tension on the slack rope *1*, etc., each rope in itself being pulled at the rate of 150 lbs., and each slack rope 75 lbs., making the table appear like this :

TABLE No. 2.—APPROXIMATE STRAINS IN THE VARIOUS ROPES WHILE TRANSMITTING 50 + H.-P., NEGLECTING FRICTION LOSSES, WEIGHT OF ROPE, AND CENTRIFUGAL FORCE. LARGE PULLEY GROOVED TO 72°, SMALL PULLEY TO 45 DEGREES.

Tight rope *1*—150 lbs.		Slack rope *1*—75 lbs.	
" *2*—150 "		" *2*—75 "	
" *3*—150 "		" *3*—75 "	
" *4*—150 "		" *4*—75 "	
" *5*—150 "		" *5*—75 "	
	750 lbs.		375 lbs.
	375 "		
	375 lbs. effective pull.		

This shows a total strain of 750 lbs. for the tight ropes, and 375 lbs. for the slack ropes, the difference being 375 lbs., which is sufficient for 50 H.-P. The maximum strain is now only 150 lbs., or one-half of that in Table No. 1. The horse-power transmitted is practically the same, and the factor of safety is about 28.

If, however, the same weight as in Table No. 1 is retained, viz., 300 lbs., the result will be that all the tight ropes will pull 300 lbs., all slack ropes 150 lbs., and the horse-power transmitted will be about 100.

Thus it appears that the effective pulling power of the large wheel is actually reduced by reducing its angular groove, and at the same time the driving effect of the transmission is nearly doubled—a sort of mechanical paradox. Assuming again that the rope is uniform, it is found that by widening the groove of the large pulley the power transmitted is actually increased.

The question naturally arises: How account for the fact, frequently observed, that it is the center strands that are usually slack? This can be explained as follows: The tension in slack rope *5* is given in Table No. 1 at 150 lbs., which may (if the power demands it) sustain $150 \times 3 = 450$ lbs. on tight rope *5*, but it will not do so until slack rope *4* is drawn out to a tension of $450 \div 2 = 225$ lbs., which is possible. If now, however, still more power is called for, then 225 lbs. on slack rope *4* may sustain a pull of 675 lbs. on tight rope *4*, which will draw up slack rope *3* to 337 lbs. This, however, is bad practice, because the maximum tension is increased far beyond 300 lbs., even in tight rope *5* pulling its maximum (450 lbs.).

On the other hand, if the loose wheel by the side of the small pulley were not used, the strain in slack rope *5* would be only 75 lbs., and the ultimate strain in tight rope *5* would be 225 lbs., and in slack rope *4* it would be 112 lbs. This shows that unless the wide angle is employed in the large pulley, it is better not to employ a loose sheave; all of which illustrates more forcibly the difficulties which accrue from the use of a uniform groove in both large and small wheels.

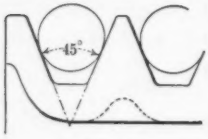

FIG. 9.

Further, it will be observed in Table No. 1 that the tension in slack rope *5* (150 lbs.) is $2\frac{1}{2}$ times as great as in tight rope *5* (60 lbs.), while with the wide angle plan, as shown in Table No. 2, the tension in slack rope *5* is half of that in tight rope *5*, as it should be.

The practical value of this system, therefore, is to increase the factor of safety, or else by employing the same factor of safety to reduce the size of rope. Thus, when the difference in the diameters is considerable, substitute $\frac{7}{8}$-in. rope where previous figuring has shown that $1\frac{1}{4}$-in. rope was required; substitute for 1-in. $\frac{3}{4}$-in., and in place of $\frac{7}{8}$-in. use $\frac{5}{8}$-in.

The reduction in the size of the rope permits also a reduction in the diameter of the pulley, and still maintains the same ratio of rope to pulley. For example, the minimum diameter of the pulley for a $\frac{7}{8}$-in. rope is 35 ins., assuming the rule of 40 diameters, while for $1\frac{1}{4}$-in. rope by the same rule a 50-in. pulley would have to be used. Then, again, the width of the sheave is thus reduced fully 25 per cent. It is not the author's intention, however, to advise employing a small

factor of safety, but, by correct figuring, to obtain the real factor of safety, which should not be less than 20.

A full discussion of this question of proportions of rope to pulley may be found in "Rope Driving" by Professor Flather.*

The author's attention was originally called to this unequal tension on the ropes of a continuous rope drive some ten years ago. At that time he had occasion to study the subject, and was much puzzled by a drive installed with wooden sheaves. This particular transmission had a driving pulley of about three times the diameter of the driven, and both wheels wore badly in the grooves. The first groove wore down fully an inch; the second, $\frac{3}{4}$ in.; the third, about $\frac{1}{2}$ in., and so on.

In another rope drive in the same building, also having wooden sheaves, but of equal diameter, the grooves wore alike. Hence, the natural conclusion, that the unequal wear was attributable in some way to the unequal arc of contact occurring in the drive with pulleys of different diameters.

The substitution of iron sheaves for wooden ones prevents this object lesson from being brought to every one's attention, but the unequal strains on the various strands are there, just as before, and frequently can be seen if one will notice the difference in the sag of the various ropes.

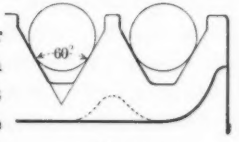

FIG. 10.

As a practical proof of the value of the variable grooves, the author would refer to a rope drive designed by him some nine years ago, and installed in the planing mill of J. K. Russell, now called the Enterprise Building, on Fulton Street, Chicago, where they have been transmitting 250 H.-P. with 12 strands of $\frac{7}{8}$-in. ropes, which lasted six years. According to the present practice $1\frac{1}{4}$-in. rope would be employed (see Fig. 7). A 14-ft. driver is grooved to an angle of 60°, and a 6-ft. driven to 45 degrees. The ropes may be seen to be pulling alike, and each is doing its own share of the work.

The calculations and tables given herein have ignored centrifugal action and journal friction, and are based on the assumption that the rope is uniform throughout, and further, that a loose wheel is provided to take the strand from the tension carriage leading from the pulling side of the driven wheel.

* "Rope Driving." By John J. Flather, Ph. B., M. M. E., page 177.

Flather's "Rope Driving"* gives a theoretical discussion of this very important matter, from which the following abstract has been prepared:

The effectiveness of the frictional grip is dependent upon the arc of contact of the rope with the pulley and also upon the coefficient of friction, which varies with the angle of the groove. The friction on each pulley will be the same if the products of the arcs of contact by the respective coefficients are equal. If the coefficient of friction of a lubricated rope on a smooth, flat metal pulley is 0.12, then, for the same rope, in a groove having an angle of θ degrees, the coefficient of resistance to slipping would be $\phi = 0.12 \operatorname{cosec} \frac{\theta}{2}$; hence, with arcs of contact of a and a' degrees the equations $\phi\, a = \phi'\, a'$, or $0.12 \operatorname{cosec} \frac{\theta}{2} a = 0.12 \operatorname{cosec} \frac{\theta'}{2} a'$ would indicate an equal frictional grip on each pulley, if it be assumed that the percentage of slip is to be the same, and that the multiplier 0.12 will give the correct coefficient of friction on each pulley.

The numerical value of the cosecant of an angle varies inversely with the angle, hence the pulley having the lesser arc of contact should have a groove with a more acute angle.

From the above equation

$$\operatorname{cosec} \frac{\theta}{2} = \operatorname{cosec} \frac{\theta'}{2} \times \frac{a'}{a}$$

in which θ and a are the angles of groove and contact, respectively, of the larger pulley; and θ' and a', similar values for the smaller pulley.

When the ratio of the arcs of contact is known and with the least angle of groove in the smaller pulley equal to 35°, 40°, or 45°, the corresponding angle of groove in the larger pulley should be as indicated in Table No. 5.

TABLE No. 5.—Angle of Groove for Equal Adhesion.

Arc of contact on small pulley / Arc of contact on large pulley $= \frac{a'}{a}$	0.9	0.8	0.75	0.7	0.65	0.6
Angle of groove in large pulley when groove in small pulley = 35°	40°	44°	47°	51°	55°	60°
Angle of groove in large pulley when groove in small pulley = 40°	45°	50°	54°	58°	64°	70°
Angle of groove in large pulley when groove in small pulley = 45°	50°	55°	60°	66°	72°	80°

* Pp. 159 to 176.

An equality of grip on each pulley may be secured by the use of an idler or binder pulley, which will increase the arc of contact on the smaller pulley, but this introduces a reverse bend in the rope, which is objectionable in rope driving and should be avoided as much possible.

Table No. 6 gives the amount of tension which may be sustained by the tight strand when the slack strand has a tension of one.

TABLE No. 6.

Degrees.	Coefficients of Friction.			Degrees.	Coefficients of Friction.		
	0.3.	0.4.	0.5.		0.3.	0.4.	0.5.
30	1.17	1.23	1.29	150	2.19	2.85	3.70
45	1.26	1.37	1.48	165	2.37	3.16	4.21
60	1.37	1.52	1.69	180	2.56	3.51	4.81
75	1.48	1.69	1.92	195	2.77	3.90	5.48
90	1.60	1.87	2.19	210	3.00	4.33	6.25
105	1.73	2.08	2.50	240	3.51	5.34	8.12
120	1.87	2.31	2.85	270	4.11	6.59	10.55
135	2.03	2.56	3.24	300	4.81	8.11	13.70

DISCUSSION.

Mr. Suplee. H. H. SUPLEE, Esq.—This paper is a very clear presentation of some facts in connection with rope transmissions, which, while they have been known for a long time, have, until now, not been used in combination or distinctly applied to the purpose for which the author uses them; at least not to the speaker's knowledge.

The English rope transmissions, using a number of independent ropes, practically depended upon the angle of the groove in the sheave to secure the necessary traction. Grooves as sharp as 30° were used at one time, but the wear on the ropes and the wedging in the grooves caused a gradual increase until it was settled by experience that from 43 to 45° was the minimum angle advisable to use in practice. It was recognized that all the ropes would not pull alike, as it was practically impossible to obtain the same initial tension, but this was expected to remedy itself from the fact that the ropes subjected to the greatest tension would stretch the most, and that this would throw a portion of the work upon the slacker ropes, and so equalize the work.

The first continuous rope transmission in the United States was probably that installed at the old shop of the Atlantic Works, in Philadelphia, about 1881, in which five turns of a $\frac{3}{4}$-in. rope were taken over a pair of 30-in. pulleys, the pulleys being of equal diameters and the ropes horizontal. The slack was taken up by an idler, and the system was identical with that now in use in many places. The sheaves were of wood (Dodge wood split pulleys), and the speaker believed that the grooves had an angle of 43 degrees. This installation was put in by Mr. Dodge, who made many experiments with grooves of different angles, even running ropes on smooth pulleys, by making enough turns to secure the necessary tractive power.

In most of these installations it was evident that all the ropes were not pulling alike, and, indeed, this must necessarily be the case if the successive action of the reduplication principle is considered. This principle of "coil friction," as Willis calls it, is well recognized, and the variation in tractive power for various arcs is correctly given by the author in Table No. 6 of his paper.

What the author has done is to pit these two sources of variation in tractive power against each other, and use the one to correct for the action of the other, and in this he has treated the subject in quite an original manner. In every case which the speaker knew, in which the angle of the grooves has been varied, the angle has been made the same in both sheaves, regardless of their relative diameter, or of their distance from each other, and therefore regardless of their different arcs of contact. The author has shown clearly how, by making the angle of the grooves correct for the difference in the arcs of contact, the tension on the different portions of the rope may be equalized,

theoretically, at least, and for this he is entitled to full and due credit. Mr. Suplee.

There is, however, another and a constructive reason for the variation in the tension of various ropes, both in the English or separate system, and in the American or continuous system, and that is the difficulty of securing or maintaining practically the same relative diameters, and hence the same velocity ratios for all the pairs of grooves. The virtual diameters of the grooves, upon which the velocity ratios of the two sheaves are dependent, are those of the circles of contact of the ropes and their respective grooves, but unless all the grooves are exactly of the same width, and the contact circles on these grooves are of exactly the same diameter, the result will be practically an attempt to drive the same sheave at various speeds at the same time. The result is a different tension on different parts of the rope, and there is a consequent wear upon the rope, and a cutting of the pulley (especially marked in wooden pulleys) which also absorbs much power. This accounts for the unequal tension in various parts of the rope in transmissions in which both sheaves are of the same diameter, and in which the arcs of contact and angles of groove should both be the same. This is a practical difficulty which can only be overcome by a much higher grade of workmanship than is usually customary with such work. When this action once commences, it continues to make itself worse, since the greater tension upon some of the parts wedges the ropes down further into their grooves and thus alters the virtual diameter of the sheave for those particular ropes and increases the variation in tension, which thus shifts from rope to rope. This, of course, has no bearing upon the merit of the author's solution of the problem as he states it, but it is a feature with which it must inevitably be complicated in practice.

For transmissions in which there is a wide difference between pulley diameters and a small distance between centers, *i. e.*, in which there is a marked difference in arcs of contact, the speaker believed that a corresponding variation in the angles of the grooves will add materially to the efficiency of the installation and the life of the rope.

H. W. Brinckerhoff, M. Am. Soc. C. E.—There is much that is interesting and valuable in this paper, but in the reasons which the author gives for the large factor of safety, from 30 to 40, now commonly employed in rope drives, a very important one seems to have been overlooked, namely, the "creep" of the rope. This creep is the slippage of the rope on the sheave as it goes off. It is caused by the change of tension due to receiving or imparting power, and is accompanied by a corresponding change of length. If, as on the driving sheave, the rope goes off at a less tension than it came on, the creep will be against the motion of the sheave; if, as on the driven sheave, these conditions are reversed, the creep will be with the motion. This can be readily studied by putting a rubber band around one Mr. Brinckerhoff.

Mr. Brinckerhoff.

forefinger and around a spool or bottle, about 1 in. in diameter, held parallel to it in the other hand, and revolving one about the other. This will give them the same relative motion as if they made the same number of revolutions per minute in the same direction, and, as their diameters are different, their surface velocities will vary, and the effect will soon appear in a marked difference of stress on the two strands of the rubber band. If now some red pencil marks have been made on the band, it can plainly be seen to contract as it leaves the spool and to stretch as it leaves the finger. This is the "creep," which is obviously inseparable from any method of driving by a band or rope having any elasticity, and it must necessarily wear the rope and the sheave or produce heat by the friction, or both, and, in any case, absorb and waste all the power that produces the wear and heat.

Fortunately, though the creep cannot be avoided altogether, it can be reduced to a comparatively harmless amount by increasing the number of driving strands, and thus diminishing the difference of tension between the taut and slack sides. For instance, if, in a given transmission, the number of strands is doubled, the average difference of tension will be halved; and, though the total length of creep on all the strands will thus be the same as before, yet, as the ropes will be less tightly pressed in the grooves, the total friction and wear will be reduced in about the proportion that the number of strands is increased. The loss of power is directly proportional to the difference of velocity between the taut and slack strands, which is, in its turn, proportional to the difference of tension between the two. Professor Unwin gives this difference of velocity at about 2% in ordinary belting, and, from the greater elasticity of ropes, it is usually much more in rope transmissions. The wear, however, varies at a more rapid ratio, for although, up to a certain point, the rope can slip in a smooth groove without serious wear, merely developing heat by the friction, yet, as soon as the safe pressure in the groove is exceeded and abrasion begins, the destruction of the rope is very rapid.

Here comes in one advantage of the wider groove on the larger sheave, as suggested by the author, for, if the groove is narrower than is needed to give the required coefficient of friction, the creep of the off-going rope takes place under greater pressure, and the rope, expanding as the tension is reduced, is actually pinched between the sides of the groove, increasing the wear and loss of power.

In view of these facts, the chief reason for the high factor of safety, stated in the beginning to be necessary for rope drives, can be appreciated. It is not because a rope drive would not be perfectly safe for a short time with a much lower factor, but because the life of such a drive would be very limited, and its waste of power would be very great. Indeed, in this case, instead of being designated a factor of safety, it would be more proper to call it the factor of economical tension.

Mr. Hoadley.

JOSEPH H. HOADLEY, Esq.—Throughout the United States, rope drives of about 700 000 H.-P., on the continuous system, have been installed. Among these there are some very interesting drives which have proved in practice that long-established theories are wrong in some details, which often happens in such cases. The wooden sheaves mentioned by the author were first adopted by the Dodge Manufacturing Company at Mishawaka, Ind., because it was thought that a rope drive would possibly have greater adhesion with a wooden than with an iron sheave, which has been practically proven in regard to the belt pulley. These wooden sheaves act exactly as the author has described them, but the speaker did not think that was caused by any unequal strain on any of the ropes. His experience had been that the grinding of the rope acts on the grooves like a file, and when the rope leaves one pulley and goes to another, as in the tension winder, there is a proportionate amount of work on the rope. For instance, a manilla rope will possibly be in use sixty days, after which, if it is taken out and examined, although it will look practically new, yet in places in the center it will be found to be ground up like flour. It is impossible to make a perfect splice after the rope has once been in use. In most cases grinding occurs where the ropes wear the pulleys, and it is worse where manilla rope is used. To avoid that trouble cotton rope has been substituted, but it has been found almost impossible to make a cotton rope of equal diameter throughout. In such case the same effect is produced as in the English drive system, *i. e.*, the ropes will not work together because the splices cannot be made of the same diameter, and one rope will creep. Even if the splices are made so that they work with strains nearly equal, the splices soon pull out, and then the creep in the rope occurs. As soon as this happens there is a greater strain on the rope from the fact that its weight and its different strands, as soon as they change their tension, create a greater driving power in one rope than in the other. As soon as that takes place, some of the ropes sink and change the pitch diameter, and this will be found in those cases where English drives with pulleys upwards of 20 ft. in diameter have been used. It can readily be seen that a change in the pitch diameter of one of the ropes, traveling so many feet per minute, means a great deal, and in that way the ropes are very easily worn out.

The drive of the Broadway Cable Road, properly proportioned, and put into a continuous rope system, should transmit about 10 000 H.-P. The engine, the speaker believes, is capable of transmitting 1 500 H.-P. The difference in the two systems can readily be seen. A continuous rope drive will drive pound per pound of rope about ten times as much as a single rope system of the old style. The single ropes, of course, have found great favor with mechanical engineers, because the rope system was first adopted in England in large drives,

Mr. Hoadley. and American engineers really secured their first information about rope drives and their installation from the English, and in that way very naturally took after them. The English rope drives in use, where the speaker employed the compound winder, run from about 500 to 4 000 H.-P. He has installed about fifty of 1 000 H.-P and about one hundred of 2 000 H.-P., and these have been in use for upwards of eight years.

The Edison Electric Light Company in San Francisco uses the compound wind system on an Allis-Corliss engine, with a 26-ft. drive wheel. There are fourteen grooves in the fly-wheel, twenty-two grooves in the driven, with 33-ft. centers; a 1¼-in. rope is used. This plant has constantly transmitted from 1 100 to 1 300 H.-P., the ropes lasting five and six years. The speaker believes that the use of the compound winder has not been appreciated by engineers. In his experience he has found that continuous drives with a steady power do not require the compound winder; but where a plant is subjected to variable powers, it is very important not to allow the rope to commence to slip. The engine wheel has the cylinder to back it up, and the travel of the fly-wheel on the engine can readily recoup lost speed; but the driven has no method of recouping its velocity other than through the transmitter. For instance, in the case of the Chicago City Railway at the Fifty-second Street Station there are five 2 000 H.-P. rope drives, or a maximum of 10 000 H.-P. The drives are fitted with a manilla rope. The fly-wheel of the engine is 20 ft. in diameter and has twenty-one grooves; the driven has thirty grooves, and on an average the dynamos connected with the engine are 33 ft. to 34 ft. The longest drive is 45 ft. The power ranges from 150 to 1 800 H.-P., and this fluctuation takes place several times during the day, owing to the heavy trains which they operate. In this plant it is found that, with the addition of the compound winder, there is practically no slip between the engine fly-wheel and the driven. The readings of the tachometer show, for 1896 or 1897, that the difference between the number of revolutions made by the engine and by the generator is practically less than 2% for the entire year, which is a very close transmission, almost as positive as cog gears. Of course, it is not as positive, because gears do not slip at all, but it can be considered as practically without slip.

There is a great deal in the way a drive is designed. The reason rope transmissions have not been more successful is that disaster has been occasioned in almost every case from either poor splicing or bad regulation of the engine. If an engine governor dances, the engine will throw the rope out of the groove very readily. In that case the rope always thrown out is that which leads from the slack part of the fly-wheel. There are cases where the winding of the rope will make a bad drive and cause a great deal of trouble. There is

only one way to have it right. If it is proportioned right, if the transmission rope is correct and the engine well regulated, it is the prettiest piece of work in mechanics. If there is anything wrong about it, it is about the worst thing that could be installed. Great care should always be taken, as far as possible, not to allow the shock occasioned by the variation of the load to drop back to the engine, and therefore metal should be put into the driven pulley to practically guard or regulate 75% of the greatest variation that can be anticipated from the maximum power of the engine. In that way a very long-lived rope and a very successful drive will be obtained. Mr. Hoadley.

In the case of steady powers, such as are found in cotton mills, the compound winder has no place, unless it is compulsory to have a very small driving pulley and very close centers, and then the contact of the driven pulley is not sufficient to permit the load to be thrown on unless it is done very carefully. In five minutes a rope can be burned to pieces, and, of course, if there is one place in it that is liable to break, it will do so and will probably wreck the plant. The speaker would rather have a rope drive for any use beyond the power of an 8-in. belt, and did not know of any case where such a drive could not be properly designed to fulfil all conditions, unless it were in the case of a very small pulley, or where the space is contracted. Of course, there is a formula that must be adhered to rigorously for large powers, and any variation therefrom will cause trouble. The speaker recently had a case, that of the Illinois Steel Company at Joliet, Ill., where a 4 000 H.-P. drive was installed for driving the rod mill, the worst character of power to which a rope drive could be put, because of the dirt in the yard and the iron filings which affect the rope injuriously and make it act all the time as though it were going to pieces. In such a case it is necessary to use a dressing which will make the heart of the rope impervious to dirt and iron filings and prevent wear from the outside, that is, prevent cutting the outside strands and starting the rope to wear.

In the case of the grooves which the author states are necessary, the speaker had found some advantage in changing in a small degree their angle when an excessively small pulley was used, and had found that the curved grooves, like the English groove, or such as are employed in drives of this character, were damaging to the rope if a splice had been carelessly made. As the rope sets down, its section will wear to an egg shape. In practice it will not wear round in any case. As soon as the rope commences to do that, it has more friction, and for that reason the speaker always uses the angle grooves instead of the hollow grooves used generally by the English.

The designing of rope drives is as much a trade as practical steam engineering, and it requires as much study. Before a man attempts to design a rope drive he should have practical experience

Mr. Hoadley. and theoretical knowledge, and be as proficient in that line as he would in starting out to design and build a steam engine. It is a business by itself. Every case that comes up has conditions entirely different from the preceding case, even though the horse-power be the same. There may be a drive of a 1 000 H.-P. with a driver running 200 revolutions a minute driving horizontal shafts, with a driven running at 400 revolutions a minute and with 30-ft. centers; and in the next drive the driver remaining the same, there may be a right-angled shaft situated on the roof, or half a mile away. All the conditions must be as carefully considered before beginning to design the drive as would be necessary in the very highest type of engineering. Many think that a rope and a groove constitute a rope drive; it does not, and that accounts for many of the failures.

The speaker had installed two drives in Paterson, N. J., for the Edison Company and could not account for the trouble experienced, as he had never seen the drives, although he had designed them. One of the ropes went to pieces about two hours after the engine started. The next rope put on has worked satisfactorily and has been absolutely noiseless. The Toledo Traction Company have four rope drives, each of about 800 H.-P. They have been in use a year and a half, and have never given any trouble. The ropes have never been replaced, and the slack has never been taken out of them. In the Illinois Steel Company's drives it was found that the wear was very rapid on account of the fluctuation in the power, and on one of the drives, the compound winder was abandoned. That was a practical test of the compound winder. The ropes ran six weeks and wore out. The tension weight required there is 2 100 lbs.; with the compound winder, the tension weight was 900 lbs. to do the same work. The speaker had carefully weighed the results and found that up to 200 H.-P. the compound winder is of no service. Beyond that, the extra adhesion necessary to prevent the commencement of slipping on the driven pulley is essential.

CORRESPONDENCE.

Mr. Webber. SAMUEL WEBBER, Esq. —The writer is pleased with the suggestions in this paper and quite agrees with the author in his proposal to enlarge the angle of the grooves in the larger pulley and so allow a sufficient slip, by which the elasticity of the ropes would allow them to equalize the distribution of the load, and bring all the chords into equal tension. The writer does not pretend to be so much of an expert on "Rope Driving" as on some other methods of transmitting power; but he has always protested against reducing the angle of the groove to less than 45° as putting unnecessary friction and wear on

the ropes; and if more power is needed than can be obtained at that angle, he would increase the number of ropes rather than reduce the angle of the groove. Neither would he increase the angle of the grooves in the larger pulley to above 65 degrees. In any case, 60° gives an equilateral triangle, in which a rope ought to free and adjust itself readily. Mr. Webber.

The actual observations, given on pages 174 and 175, of the different wear of the grooves in wooden pulleys when the angle in both cases was 45°, are very conclusive and satisfactory. The case quoted on page 170 seems to show that the continuous failure of the ropes was due to the small diameter of the smaller pulley, which was scarcely 19 diameters of the rope, and is confirmed by the writer's observations in two similar cases, where he was called in to examine and advise after the mischief was done. He would never advise less than 40 diameters, or perhaps 36 for cotton rope, if he expected the ropes to do satisfactory service. The case of Mr. Russell's mill in Chicago, quoted on page 175, seems very satisfactory proof of the advantage of the plan of increasing the angles of groove on the larger pulley.

The rough sketch from an English "template" (Fig. 11), which the writer has carried in his pocket these ten years, shows what seems to him a very satisfactory form of groove.

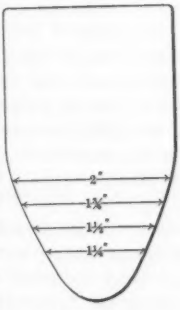

FIG. 11.

J. J. FLATHER, Esq.—The author has brought out very clearly the advantages of using a larger angle in the groove of the pulley which has the greater arc of contact. Although this construction has been used for a number of years, its application is not as general as its advantages would warrant. Not only is the load more evenly distributed, but, as will be noted in the analysis of the stresses, the total load may be twice as great for a given allowable stress as that obtained with similar pulleys having equal angles in the groove. Mr. Flather.

With a continuous rope drive, such as here indicated, the use of a loose idler adjoining the smaller pulley, first introduced by the author, is an advantage in every case, for momentary fluctuations are constantly occurring, and the tension carriage, unless loaded unnecessarily, will be moving in or out continually, thus causing the rope to be dragged through the groove of the driven pulley unless a loose idler be used, in which case simple rotation is given to the idler.

While agreeing with the author as to the advisability of using grooves having different degrees of inclination when the diameters of the pulleys differ appreciably, the writer would go further and recommend that the sides of the grooves be arcs of circles having radii proportional to the diameters of ropes to be used and to the variable angle

Mr. Flather. desired, as shown in Figs. 12 and 13. Although the tension in each member of a rope drive may be assumed to be the same, there is unquestionably a variation of pull due to momentary fluctuations, which must either be absorbed by the elasticity of the rope or transmitted through it until the strains are equalized. If the rope used is uniform in structure, that portion which receives the greater strain will be drawn more deeply into the groove: and if the latter be of the curved form, the coefficient of friction will be reduced; so that the resultant adhesion produced in this portion of the rope will be less than that in some other part which, under a lighter strain, occupies such a position on the pulley that its coefficient of friction, and consequent adhesion, is greater for a given back-tension.

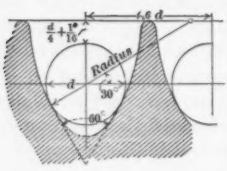

FIG. 12.

With the usual arrangement of pulleys, namely, that in which the larger wheel is the driver, the tendency of an increase in tension is to increase the velocity of the driven pulley. If several wraps of a series occupy positions in the grooves such that with a given ratio of pulley diameters the velocity of the smaller pulley is twice that of the driver, any decrement, x, of the effective radii of another wrap, which is drawn more deeply into the grooves of the two pulleys, will alter the velocity ratio from $\frac{2}{1}$ to $\frac{2-x}{1-x}$. The tendency of this wrap, then, is to produce a greater velocity in the driven pulley, which cannot occur without some slip in the other wraps; but these wraps have a greater adhesion, and therefore tend to drive the pulley at a less number of turns per minute, which will produce slip in the more heavily strained member; hence the effect of any change in position due to sudden increased strain on one wrap will tend to quickly adjust the tensions in all portions of the rope and neutralize any inequalities in driving effort. With continuous rope transmissions this effect will not be so apparent, although it exists to a certain extent on account of inequalities in the rope and various mechanical imperfections in the system.

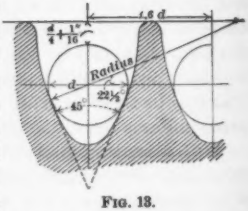

FIG. 13.

In his paper the author has used a pulley diameter equal to forty times the diameter of the rope. This will be found very satisfactory for general work, but if, for any reason, it is necessary to determine the least diameter of pulley which may be used in a given case, the speed of the rope should enter as a factor, since the wear and consequent life of the rope will vary with its velocity. The flexibility of

the rope also enters into the problem, since this varies inversely as the cube of its diameter. Other considerations would indicate that while forty times the diameter of rope may be satisfactory for a 2-in. rope running at a high velocity, it is also true that a lesser proportion will give equally satisfactory results for a smaller rope or for a case where the speed is not so great. Mr. Flather.

SPENCER MILLER, M. Am. Soc. C. E.—Friction plays the important part in rope-driving, and the coefficient varies with the temperature, with the degree of lubrication and with the degree of finish of the sheaves. Centrifugal force, "creep," weights of rope, all are elements which make the behavior of the strands of a rope in a continuous rope drive irregular. Any attempt to figure strains must, therefore, be an approximation. Mr. Miller.

On account of this fact the author desires to express his appreciation of the expressions of approval of his paper from those who have taken part in the discussion and who have confined their comments to the problem taken broadly.

Answering Mr. Webber, the author would say that the rope drive taken for illustration now has separate cotton ropes, and the dynamo is mounted so as to slide and thus take up the stretch of the ropes. When one of the ropes breaks, all are discarded, and new ones of exactly the same length and from the same coil are put on. The drive is so small that this is permissible. The ropes last many times longer than before and are entirely satisfactory, which would seem to prove that when all the ropes pull alike, satisfactory results are obtained.

Mr. Flather's discussion is very interesting, but the author does not agree with him as to the use of curved-sided grooves for continuous rope driving. A continuous rope may be made so even in density that it may be depended upon for uniformity as to its diameter, excepting where the splice occurs.

A groove with sides curving toward one another, as shown in Figs. 12 and 13, interferes with the free passage of the splice. The splice is larger and weaker than any other part, and should be allowed to pass easily. The more the sides curve in above the line of normal contact, the sharper the angle, and the splice in passing rides higher in the groove than the other strands. The splice will move slower than the portion of the groove with which it is in contact, and as that part of the groove is less polished and of a sharper angle, it stands to reason that the splice will slip and wear badly. This argument, however, applies to drives where four or more wraps of rope are used. The author would prefer a groove with sides curving away from each other, and having an angle of 45° where the rope is in contact when worn to its minimum diameter.

As the rope wears smaller (and hence becomes weaker), it rests in a wider angle than when new (using the curved form of groove shown in

Mr. Miller. Figs. 12 and 13), and each strand must pull more to transmit the same horse-power, as will be seen by calculation.

Roughly speaking, the ratio of the slack to the tight rope on 60° is 1: 2 (180° arc of contact) while on 45° it is 1: 2½; hence, an effective pull of say 150 lbs. with an angle of 45°, can be secured with 250 lbs. tight strain and 100 lbs. slack strain; while on 60° it would require 300 lbs. tight strain and 150 lbs. slack strain. Hence it is seen that the strain is increased 20 per cent. The journal friction is increased 30%; for the sum of the tension, with an angle of 45°, is only 350 lbs., while with an angle of 60°, it is 450 lbs.

The straight-sided groove is cheaper to make, easier to caliper, and for continuous rope driving it is preferred by the author to that shown in Figs. 12 and 13.

Mr. Brinckerhoff defends factors of safety of 30 to 40, because of wear of rope due to "creep." To show that such factors are not essential to the satisfactory life of the rope, the author would refer to the life of the rope in the Russell plant, mentioned on page 175, and add the following data recently secured.

The driving engine—a Hamilton-Corliss, 22 x 48 ins.—will develop, according to the makers, 270 H.-P. at 70 revolutions, with 80 lbs. steam pressure and one-quarter cut-off, and these are the conditions under which the plant has been running. The fly wheel was grooved so as to take any rope from ⅞ in. up to 1¼ in. The first rope put on was ⅞ in. and lasted six years. This was a common tallow laid rope, of probably not over 6 000 lbs. ultimate strength. The speed of this rope is 3 080 ft. per minute, and the effective pull 2 680 lbs.; or, assuming a ratio of 2 to 1, the pull on each tight rope would be 446 lbs., showing that a factor of safety of only 14 was sufficient for a life of six years. However, 1¼-in. rope is now in use, and supplies a factor of safety of 20, which is as small as should be used.

The author would reiterate that if a factor of safety of 20 is actually obtained, taking into account all strains, it will be satisfactory. In many instances, however, he would urge a higher factor.

If the life of the rope was the only consideration, then U-grooves of highly polished iron would accomplish the result, but at a tremendous loss in journal friction. A midway position, avoiding excessive wear of rope on one side, and excessive journal friction on the other, the author believes to be in an angle of groove of about 45° in the pulley of the less arc of contact, and in widening the other to the limit proposed.

The weight of the rope must always be taken into account, however, for in vertical drives the weight of the rope increases the hold on the upper wheel, and, if sufficiently high above the lower, may have (in spite of its smaller diameter) a wider angle than the larger pulley.

So far as the author knows, Mr. Hoadley is the only other engineer who has attempted to solve the problems occurring in rope

drives where there is a considerable difference in the diameters of the driving and driven pulleys. His compound winder has been extensively employed, and the results obtained are considered generally satisfactory. If the rope drive which has been used as an example had been remodeled by the introduction of a compound winder, the drive would appear as in Fig. 14, in which it will be noticed that the tension wheel is horizontal and above the plane of the driving and driven wheels. A winding pulley is placed between the pulleys and to one side of the driven pulley, and the driving rope makes two extra turns around it and the driven pulley before it goes to the tension carriage. Mr. Miller.

The strains in Table No. 7 may be seen in this example to be on an ascending scale, gradually multiplying from the last rope to the first, the compound winder serving to let down the strain engendered in the first rope to that of half the tension weight.

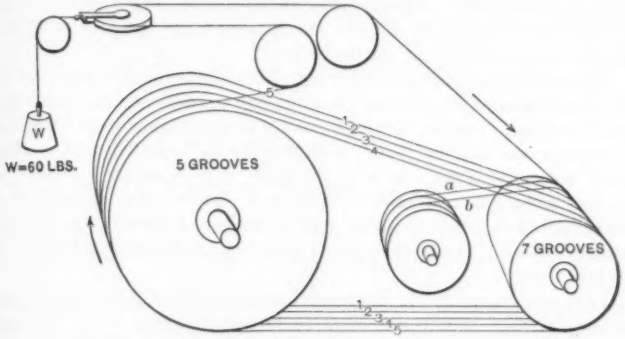

FIG. 14.

Table No. 7 illustrates the strains in the drive when a compound winder pulley is used, both wheels grooved to 45°, and transmitting the maximum of power, with 300 lbs. as the maximum strain on the driving rope, neglecting the strains on the ropes to the winder pulley.

TABLE No. 7.—APPROXIMATE STRAINS WHEN THE COMPOUND WINDER PULLEY IS USED; BOTH WHEELS GROOVED TO 45 DEGREES.

Tight rope, 5.....	60 lbs.		Slack rope, 5.....	30 lbs.	
" 4.....	90 "		" 4.....	45 "	
" 3.....	135 "		" 3.....	68 "	
" 2.....	204 "		" 2.....	102 "	
" 1.....	306 "		" 1.....	153 "	
	795 "			398 "	

$$795 - 398 = 397 \text{ lbs. effective pull.}$$

$$\frac{397 \times 4\,518}{33\,000} = 54 \text{ H.-P.} +$$

Mr. Miller. Thus, it appears that with the same maximum strain (about 300 lbs.) the total power transmitted is about the same as when the winder pulley is omitted, but it is also noted that while it is impossible with 60 lbs. tension to carry the strain beyond 306 lbs., in the previous example (Fig. 7), both wheels grooved to 45°, a tension weight of 300 lbs. made it possible to pull as high as 675 lbs. on tight rope No. 4.

The author believes that the real virtue in the compound winder is the fact stated by Mr. Hoadley, that a sudden application of load will not cause slip on the small pulley any more than it would on the large pulley, and, furthermore, that there can be no sudden increase in any strand beyond its safety limit.

But, with the author's plan, whereby the resistance to slipping is the same in both pulleys, there is no more chance of the rope slipping on the small pulley than on the large. The author's plan also prevents a sudden rise in the strain on an individual strand as well as Mr. Hoadley's (see Table No. 2), but the ropes do not pull alike in the latter, and they do in the author's plan.

The author has seen many of the plants designed and installed by Mr. Hoadley, and all of them show by the relative sag in the ropes that the strain gradually lessens exactly in accordance with Table No. 7, but to a less degree. When a rope drive is installed with pulleys near together, and the rope is quite tight, a very slight difference in sag may mean a considerable difference in strain. For example, in a drive with 34-ft. centers, using a $1\frac{3}{4}$-in. rope, a difference of $\frac{1}{4}$ in. between a sag of $1\frac{1}{4}$ ins. and $1\frac{1}{2}$ ins. represents a difference in tension of 220 lbs.

The example given of the Russell mill drive is a good test of the author's method, for it was used to transmit power for a saw and planing mill, and the sudden applications of load in such a factory are fully as great as in driving an electric plant.

The author agrees with Mr. Hoadley in the file-like action of manilla rope and wooden sheaves, but the irregular wear must be attributable to unequal strains in the strands.

Rope driving has certainly made great progress in the United States, but, unfortunately, a rope drive is easier to put up than belts and pulleys, and many installations have been made with the single idea of driving the wheels and transmitting the requisite power.

In consequence of this there has been a great deal of shiftless rope driving which has done much to prejudice engineers and mill owners against its use.

The author believes then that it is most important to keep in mind in designing a continuous rope drive that each strand must feel the effect of the tension weight equally, and that the holding power of the rope to the driving wheel (all things taken into account) must be exactly equal to that on the driven wheel.